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A COMPARISON OF PREDICTED AND OBSERVED GAS TURBINE PERFORMANCE

CORLISS WYATT ADAMS
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A COMPARISON OF PREDICTED AND OBSERVED GAS TURBINE PERFORMANCE

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SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS
FOR THE DEGREE OF MASTER OF SCIENCE
IN AERONAUTICAL ENGINEERING

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY
(1952)

ABSTRACT

A COMPARISON OF PREDICTED AND OBSERVED GAS TURBINE PERFORMANCE

by

Corliss Wyatt Adams
and
Richard Miles Shively, Jr.

Submitted to the Department of Aeronautical Engineering on May 16, 1952, in partial fulfillment of the requirements for the degree of Master of Science in Aeronautical Engineering.

The purpose of the investigation was to compare predicted and and observed gas turbine performance. It was also desired to study the effect of Reynolds Number in the performance characteristics of a gas turbine.

A Westinghouse X9.5B Aviation Gas Turbine was modified by the authors and used to obtain the observed data. The predicted performance was determined by means of the C. Richard Soderberg method of Loss Coefficients for Turbine Passages.

It was determined that the observed efficiency was 74 per cent as compared with a predicted efficiency of 86 per cent. Due to the failure of the turbine midway through the data runs, no conclusions could be made regarding the effect of Reynolds Number on gas turbine performance.

The investigation was conducted by the authors at the Gas Turbine Laboratory, Massachusetts Institute of Technology, Cambridge, Massachusetts.

17317

Cambridge, Massachusetts
May 16, 1952

Professor Shatswell Ober, Chairman
Departmental Committee on Graduate Students
Department of Aeronautical Engineering
Massachusetts Institute of Technology
Cambridge, Massachusetts

Dear Sir:

In partial fulfillment of the requirements for the degree of Master of Science in Aeronautical Engineering, we herewith submit our thesis entitled, "A Comparison of Predicted and Observed Gas Turbine Performance."

Respectfully,

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I INTRODUCTION

The purpose of this thesis was to compare a calculated turbine performance with an actually tested turbine performance. The method for calculations was that of Professor C. Richard Soderberg of the Massachusetts Institute of Technology. This method involved the calculation of loss coefficients and included corrections for Reynolds number, aspect ratio, and inlet air angles other than nominal. Tip loss corrections were also made. It was desired to use the experimental data to check the accuracy of the Soderberg curves.

It had originally been intended to study Reynolds number effects on the turbine performance by varying the turbine inlet density, but due to the breakdown of the turbine, insufficient data was obtained to accomplish this.

This investigation was performed by Richard M. Shively, Jr., and Corliss W. Adams during the academic year 1951 to 1952 in the Gas Turbine Laboratory of the Massachusetts Institute of Technology, Cambridge, Massachusetts.

The authors wish to express their gratitude to Professor E. S. Taylor, Associate Professor E. P. Neumann, Assistant Professor R. C. Dean, Mr. James F. Hands, and Mr. Dalton Baugh for their generous assistance and cooperation.

II EQUIPMENT

The investigation was conducted with a modified X9.5B Westinghouse Gas Turbine Jet Aircraft Engine mounted in a compressor test stand designed and built in 1951 by Padis, Swainson and Gern. The turbine was operated cold and driven by air supplied by the Sloan Gas Turbine Laboratory wind tunnel.

The jet engine was so modified as to isolate the closed air systems of the turbine and compressor. This was accomplished by means of an air tight 1/8 inch steel diaphragm inserted in the combustion chamber. See Fig. 1 . The air from the wind tunnel system entered the turbine via a steel annulus built around the combustion chamber and exhausted through the tail section of the turbine into the low pressure side of the tunnel system. The tail section was utilized because it presented a straight section of no area change for outlet measurements.

Because of the closed cycle system used in the wind tunnel, it was necessary to modify the bearing lubrication system. Under normal operation the bearings received a lube oil mist from the oil pump and air mixer and discarded the excess oil with the exhaust gases. In order to make this correction, Padis, Gern, and Swainson designed a lube oil seal which fitted over the cooling fins behind bearing No. 3. See Figs. 2&3. Air at 40 psi was supplied to the seal to prevent leakage of oil into the turbine. The excess oil from the bearings collected in the bottom of the combustion chamber outer casing and was evacuated by a connection to the laboratory steam ejector. It was later found that the lube oil seal did not function perfectly. A considerable quantity of the oil escaped through the lube oil seal and into the exhaust of the turbine. As will be shown in the results, this

failure tended to falsify the data.

The compressor of the jet engine served as a variable load for the turbine and was controlled by means of a 10 inch gate valve behind the compressor, a connection to the steam ejector, and a 3 inch valve to admit ambient air. The air in this system was cooled by means of two water coolers.

The wind tunnel used to supply air to the turbine consisted of a large 5 stage compressor and several steam ejectors in addition to a by-pass valve. Inlet air conditions to the turbine were controlled by the rpm of the wind tunnel compressor, the temperature of the cooler, and the wind tunnel ejectors. Fig. 4 is a schematic diagram of the wind tunnel system.

The instrumentation of the turbine consisted of static pressure taps and total head tubes at the entrance to the nozzle block, pitot static tubes at the exit of the turbine and total temperature thermocouples at the entrance and exit of the turbine. In order to determine mass flow, a pitot static tube and thermocouple were inserted in a straight 20 foot section of the wind tunnel piping. Bearing temperatures were carefully watched by means of thermocouples furnished as standard equipment with the engine. The speed of the turbine was obtained from a standard aircraft engine tachometer. For further details regarding the instrumentation refer to Appendix A.

III PROCEDURE

The variables available in the test procedure were considerable in number. In the wind tunnel system the density of the inlet air to the turbine was varied by means of the compressor rpm, the cooler temperatures, and the setting of the steam ejectors. The load of the jet engine compressor was controlled by the gate valve, the ambient air valve and the steam ejector. In connection with the compressor air circuit, it was discovered that there was considerable leakage of air into the turbine system through the bearing housing, the lube oil seal, and into the space between the nozzle block and turbine wheel. This deficiency in the test set-up provided an additional source of experimental error.

The procedure consisted of runs made at various compressor loadings to give a performance curve for a particular rpm. The load on the compressor was first set and the turbine rpm returned to the desired rpm by adjustments of the wind tunnel compressor speed and inlet air density. In order to keep the performance runs on a comparable basis, it was necessary to maintain the inlet density within a ten per cent (10%) range. The compressor loadings were so selected as to give a maximum and minimum pressure ratio across the turbine on the initial runs. This gave an indication of the available load range. Data at one or two intermediate points were then taken in order to give a satisfactory curve. This same procedure was followed in obtaining data for other rpms.

Mass flow data for each run was obtained from pressure and temperature readings taken in the wind tunnel piping mentioned above. A

standard pitot-static tube was installed in such a manner as to permit traversing the 12.6 inch pipe to obtain pressure readings at four (4) radii of equal areas. A total temperature reading was obtained from a copper-constantan thermocouple installed near the pitot static tube.

From the data of the preliminary runs, a relationship was developed to give mass flow directly. The pitot-static was located at the mid position or center of the pipe and left there for all remaining runs. Since the velocity of the air in the pipe at measuring point was always less than 100 ft. per sec., we can assume incompressible flow. From the continuity equation

$$\dot{m} = \rho V A$$

and Bernoulli equation

$$p_0 = p + \frac{1}{2}\rho V^2$$

we obtain

$$\dot{m} = \sqrt{\frac{2p\,\Delta p}{RT}}\cdot A = 11.0\sqrt{p\frac{\Delta p}{T}}$$

where

m = mass flow in pounds per second
p = static pressure in inches of Hg.
Δp = difference between static and total pressures in inches of water
A = derived constant which includes unit conversion factors and cross sectional area of pipe
T = static temperature in degrees Rankine

The pressure data for mass flow was obtained from readings given by a manometer board shown in Plate A. The instrument in the upper left hand corner of the board is an inclined differential manometer to give readings of Δ p.

IV RESULTS AND DISCUSSION

The experimental results of this experiment are shown in Fig. 11. All data is on file at the Gas Turbine office of M.I.T. Figure 11 is a non-dimensional plot of mass flow vs. total pressure ratio across the turbine, for different rpm's and at an inlet density constant within ten per cent. The inlet density used was approximately the inlet density to the turbine when run as a jet engine at rated power at sea level. The turbine efficiency is marked beside each data point. It was the original intention to obtain the same plot for other inlet densities, but due to the breakdown of the turbine, this was not accomplished. The non-dimensional mass flow parameter, $\frac{\dot{m}\,N}{750\,p_{o1}}$, is proportional to $\frac{\dot{m}}{\dot{m}^*} \times \frac{N}{\sqrt{\theta}}$.

where: $\dot{m} =$ air mass flow, lb/sec

$\dot{m}^* =$ maximum mass flow through area whose diameter is the wheel diameter, for the inlet total pressure and temperature

$\dot{m}^* = \frac{.53\,p_{o1}}{\sqrt{T_{o1}}} \times \frac{\pi D^2}{4}$ for $k = 1.4$

$N =$ revolutions per minute

$\theta = \frac{T_{o1}}{540^\circ R}$

The factor $\frac{N}{\sqrt{\theta}}$ was used simply to spread the lines apart.

It was also intended to obtain data over a wider pressure ratio range at each rpm, but this was impossible because of the limited range of the compressor load. Each rpm line is from the minimum load to maximum load attainable.

Before discussing the results, an over-all picture of the measurement errors will be given. The inlet pressure readings were inaccurate due to extreme turbulence, which was caused by the sharp right angle turn to the

turbine inlet. The inlet static pressure readings were worthless, there being fluctuations in the mercury column as much as three inches during some runs. The fluctuations for the inlet total pressure were far less, but an accuracy closer than one-half inch of mercury cannot be assumed. There were no fluctuations in the outlet pressure readings.

The sources of error in temperature readings were as follows°

1. Insensitivity of potentiometer. This permitted an accuracy to within 2 to 3°F, either high or low.
2. Warm air from compressor circuit. Caused high reading for turbine outlet air temperatures.
3. Warm oil from bearings. Caused high reading for turbine outlet air temperatures.
4. Air yaw angle more than 30° to turbine outlet thermocouples, for some runs. All the total temperature thermocouples were located for axial flow, and it is believed that for some runs the turbine outlet velocity was more than 30° from the axial direction. According to reference (a) the recovery factor falls off sharply for yaw angles greater than 30°.

Temperature errors due to heat loss, conduction, or radiation were negligible.

Referring again to Fig. 11, the curves appear to be consistent in respect to mass flow vs. pressure ratio.

There are several inconsistencies in the trend of the efficiencies. The value of the efficiencies are extremely critical to variation in temperature readings, for the relatively low powers involved in this experiment. For example, consider point A of Fig. 11. A variation of two degrees Fahrenheit of the outlet temperature will change the efficiency more than five per cent. A variation of inlet pressures will also change

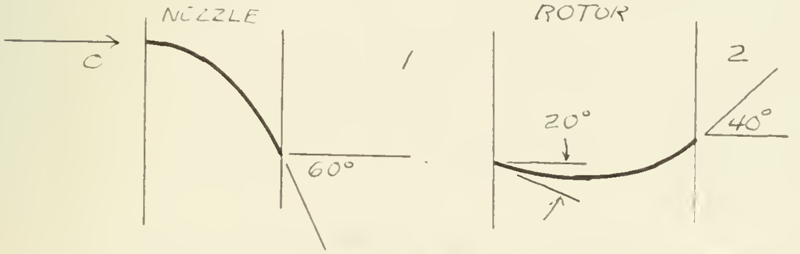
the efficiency, but not as much. For the point in question, a variation of one-half inch of mercury for the inlet total pressure will change the efficiency about 2 1/2 per cent. It is indicated that the inconsistencies in the trend of the efficiencies were due mainly to errors in temperature readings.

In addition to the inconsistencies in the trend of the efficiencies, the actual efficiencies were much lower than expected or calculated. The possibility that the outlet temperature readings were higher than the actual temperatures could account for this.

It is recommended for turbine tests with a small temperature drop across the turbine, that a device such as a dynamometer be used to measure power, rather than using temperature difference. This would also permit a wider variation of load for the turbine. It is further recommended that provision be made for a smooth air flow into the turbine in order to obtain accurate inlet pressure readings.

The second purpose of this thesis was to check observed performance with a calculated performance. Turbine dimensions were obtained by measurement. The blade angles were measured at the mid-radius only, and all calculations were made for the mid-radius only. One data point was checked by the calculations. This point is marked A on Fig. 11, and was chosen because it had the best efficiency on the rpm curve closest to design rpm.

The blade angles, notation, and initial conditions were as follows:



$p_{oo} = 30.2''$ Hg.

$T_{oo} = 537^{o}R$

$\dot{m} = 4.63$ lb/sec

Calculations were made as shown in "Sample Calculations".

Upon completion of the calculations for these given conditions, it was found that the total pressure ratio across the turbine was 1.27, whereas the total pressure ratio for the experimental run was 1.46. In order to compare the calculated work and efficiency with the observed, it was necessary that the total pressure ratios agree. Calculations were then made for other nozzle exit blade angles, keeping the other given conditions constant, and the curves of Fig. 12 were obtained, which show the effect of nozzle exit blade angle on the total pressure ratio and work. It is noted that the effect of nozzle exit blade angle on the pressure ratio and work is extremely critical. A nozzle exit blade angle of 62.4° was found to correspond to a pressure ratio of 1.46. This gave a calculated work of 11.1 Btu/lb vs. the observed work of 9.82 Btu/lb.

To find the effect of mass flow change on work and pressure ratio, calculations were made to plot Fig. 13, for a nozzle exit blade angle of 60°, and other initial conditions the same as before. To obtain the pressure ratio of 1.46, a 7.4% increase in mass flow was required. Thus, it is seen that pressure ratio and work are sensitive to mass flow change, but not as sensitive as to nozzle exit blade angle change.

Similar calculations were made to find the effect of rpm change on pressure ratio and work. These curves are shown in Fig. 14. The pressure ratio and work are relatively insensitive to rpm change.

It is to be noted that for the data point in question, the turbine nozzle is not choked, but is just short of it.

A practical result of the above calculations is that for an actual gas turbine, in the unchoked region, a small error or deviation in the nozzle exit blade angle can cause a considerable change in the mass flow. It also indicates that a prediction of turbine performance from measured blade angles is difficult.

In order to compare the calculated efficiency with the experimental

efficiency, it was decided to use the experimental mass flow and rpm as for pt. A of Fig. 11, and to use the revised blade angle of 62.4°. This was done since it was definitely more probable that the mass flow was correct within 7.4% than that the measured blade angle was correct within 2.4°.

Efficiency calculations were made using the Soderberg charts, Fig. 10, which include corrections for aspect ratio, Reynolds number, and inlet angles other than nominal. To account for tip losses, an arbitrary 4% loss in stage efficiency was applied. This is in accordance with the method of W. R. Hawthorne (ref. d). The resultant calculated turbine efficiency was 86% as compared to the experimental efficiency of 74%. This was as expected and the reasons for the low observed turbine efficiency have been given earlier in this section.

The turbine breakdown was apparently caused by a failure of the nozzle block. Upon investigation it was found that all the nozzle blades were missing from the turbine, and the outer half of all the rotor blades were chewed off. The probable cause of the nozzle block failure was that one of the nozzle blades became loose and came in contact with the wheel, causing all the nozzle blades to be wiped out. The nozzle blades apparently had received rough treatment during installation of the steel diaphragm in the combustion chamber, and one blade in particular was noted to be quite loose. If such were the case, it is indicated that similar trouble would not be encountered in other engines of this type.

V CONCLUSIONS

A calculated efficiency of 86 per cent was obtained by the Soderberg method to compare with an experimental efficiency of 74 per cent. The observed efficiency is believed low due to unreliable temperature readings.

VI RECOMMENDATIONS

It is recommended for turbine tests with a small temperature drop across the turbine, that a device such as a dynamometer be used to measure power, rather than using temperature difference. This would also permit a wide variation of load for the turbine.

Provision should be made for a smooth air flow into the turbine in order to obtain accurate inlet pressure readings.

If turbine outlet temperature readings are used, a positive means of preventing lubricating oil leakage into the air circuit should be provided.

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APPENDIX A

INSTRUMENTATION

The instrumentation for this investigation used standard type static and pitot-static tubes for pressure measurements and copper constantan thermocouples for temperature measurements.

The static pressure tubes, three (3) in number, were located ahead of the turbine as shown in Fig. 3. They were constructed of one-sixteenth (1/16") inside diameter steel tubing and inserted in the nozzle ring at 120° positions. Considerable care was taken to insure smooth surfaces where they opened to the air stream. The total head tubes used in the inlet were constructed of a small guage hypodermic tubing .05" in diameter. They were located in approximately the same plane as the static tubes and were free to move in yaw and radial traverse. Installation was accomplished by means of air-tight brass compression fittings.

For turbine outlet pressure measurements, two standard type pitot-static tubes were installed in the tail cone section as shown in Figs. 3 and 6. Brass compression fittings were used to secure them to the tail section and permit positioning in radial traverse and in yaw. All pressure readings were taken from a mercury manometer board.

The thermocouples were cylindrical half shielded total temperature probes using 28 gauge copper and constantan wiring. The details for their construction were obtained from Pratt and Whitney Report on Cylindrical Total Temperature Probes (PWA 559). Fig. 7 shows head-on and cut-away views of the probes. This type of thermocouple is relatively insensitive to orientation up to an angle of 35° as shown in PWA 559. It is noted that beyond this angle the recovery factor falls

off sharply.

The turbine outlet thermocouples were located in the tail cone at 120 positions and at different radii. This arrangement permitted a comparison of uniformity of temperature readings. The inlet to the turbine thermocouple was of the same type as described above and was located in the inlet piping to the annulus.

Wires from the several thermocouples were led to a selector switch mounted on an instrument board. A thermos bottle filled with ice and water served as a reference point for the temperature measurements and was located under the instrument table.

The thermocouple readings were obtained from a Rubicon Portable Potentiometer, Catalog No. 2700. The instrument operated in a satisfactory manner in every respect but one. It did not give as sensitive a reading as it should have been expected based on the subdivisions of its dials. For example, it was impossible to obtain a reading closer than one millivolt; i.e., the galvanometer reading would show no deflection over a range of that magnitude. As experience was gained in its operation and characteristics, the spread of temperature readings was reduced materially.

APPENDIX B

SAMPLE CALCULATIONS

All calculations were based on k = 1.4

A. Experimental data calculations for Point A of Fig. 11.

1. Notation:

1 -- turbine entrance.

2 -- turbine exit.

2. Determination of mass flow.

After several runs it was found that for the pitot tube in the center of the wind tunnel pipe, the mass flow could be obtained by the formula:

$$\dot{m} = 11.0\sqrt{p\frac{\Delta p}{T}}$$

where: p = static pressure in centimeters of mercury.

Δp = difference between static and total pressure in inches of water.

T = temperature in degrees Fahrenheit.

For point A:

$$\dot{m} = 11.0\sqrt{\frac{80.9 \times 1.2}{538}} = 4.68\ \frac{lb}{sec}$$

3. Turbine efficiency.

$$\eta_t = \frac{1-\frac{T_{02}}{T_{01}}}{1-\left(\frac{P_{02}}{P_{01}}\right)^{\frac{k-1}{k}}} = \frac{1-\frac{496}{537}}{1-\left(\frac{20.6}{30.2}\right)^{.286}} = .740$$

4. Work $= h_{01} - h_{02} = 128.34 - 118.52 = 9.82\ \frac{Btu}{lb}$

B. Predicted performance calculations.

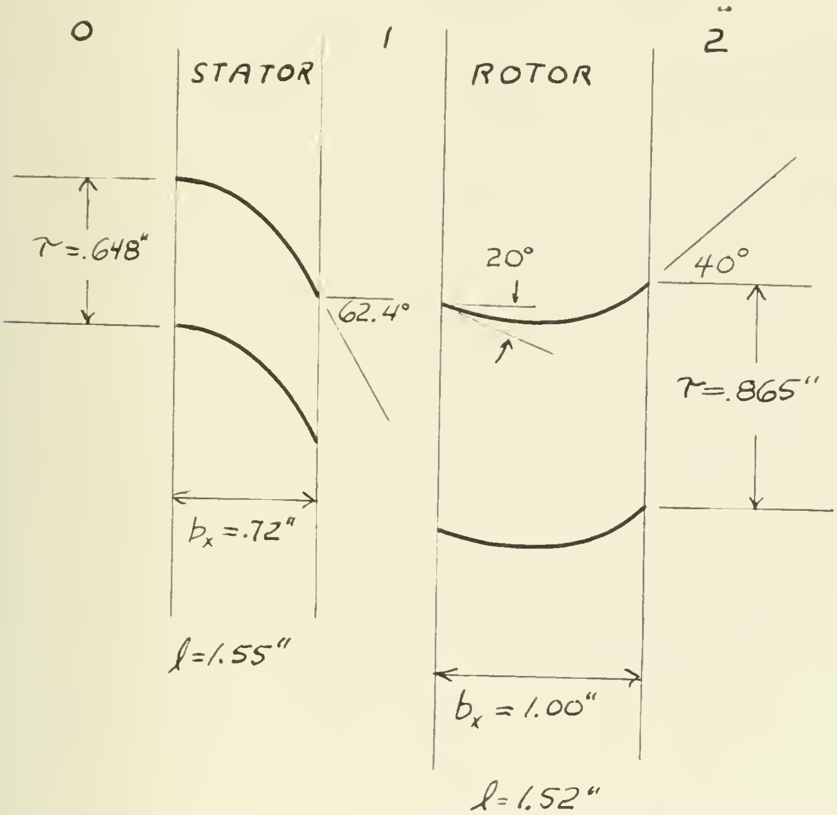
1. Notation:

0 -- Nozzle entrance.

1 -- Nozzle exit and wheel entrance.

2 -- Turbine exit.

2. Determination of total pressure ratio across turbine and work of turbine for the following inlet conditions and blade geometry: (Ref. Fig. 10)



$P_{00} = 30.2'' \, Hg$

$= 2135 \, \frac{lb}{ft^2}$

$T_{00} = 537°R$

$\dot{m} = 4.68 \, \frac{lb}{sec}$

RPM = 22,300

$u_m = 642 \, \frac{ft}{sec}$

(blade speed)

Annulus Area $= 32 \, in^2 = .222 \, ft^2$

$$D_{h_s} = \frac{2 \times .648 \sin 27.6° \times 1.55}{.648 \sin 27.6° + 1.55} = .503'' = .0419 \, ft$$

$$D_{h_r} = \frac{2 \times .865 \sin 50° \times 1.52}{.865 \sin 50° + 1.52} = .922'' = .0768 \, ft$$

a. Find inlet Mach number, M_o.

$$\dot{m} = \frac{.53\, p_{oo}}{\sqrt{T_{oo}}} A^* \quad , \quad \frac{\dot{m}}{A^*} = \frac{.53\, p_{oo}}{\sqrt{T_{oo}}}$$

$$\frac{\dot{m}}{A^*} \times \frac{A^*}{A_o} = \frac{.53\, p_{oo}}{\sqrt{T_{oo}}} \times \frac{A^*}{A_o}$$

$$\text{so} \quad \frac{A^*}{A_o} = \frac{\dot{m}}{A_o} \frac{\sqrt{T_{oo}}}{.53\, p_{oo}} = \frac{4.68\sqrt{537}}{.222 \times .53 \times 2135} = .431$$

$$\frac{A_o}{A^*} = 2.32$$

From the air tables for isentropic one dimensional flow,

$$M_o = .26$$

b. Use of polytropic charts to obtain the velocity triangles and temperatures.

(1) Nozzle.

From previous calculations, it was estimated that $\zeta_s \cong .065$

$$\therefore \eta_{ep} = \frac{1}{1+\zeta_s} = \frac{1}{1.065} = 94\%$$

From polytropic chart, $n = 1.37$

Enter with $M_o = .26$, then $\frac{A^*}{A_o} = .45$

$$\frac{A^*}{A_1} = \frac{A^*}{A_o} \times \frac{A_o}{A_1} = .45 \frac{\cos 0^\circ}{\cos 62.4^\circ} = .972$$

$$\therefore M_1 = .77 \,, \quad T_1 = \frac{T_{o1}}{1+.2M_1^2} = \frac{537}{1.1186} = 480^\circ R$$

$a_1 = 49.1\sqrt{T_1} = 1075 \frac{ft}{sec}$

$c_1 = M_1 a_1 = 828 \frac{ft}{sec}$

$c_{x_1} = 828 \cos 62.4° = 383.5$

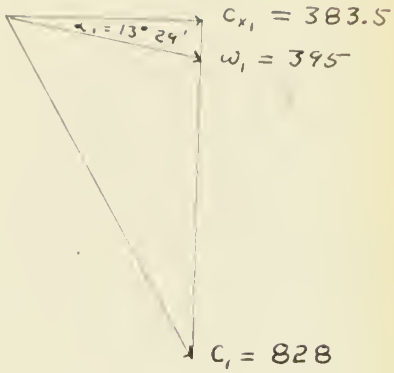
$c_{u_1} = 828 \sin 62.4° = 734$

$\omega_{u_1} = c_{u_1} - u$

$= 734 - 642 = 92$

$\tan \alpha_1 = \frac{\omega_{u_1}}{c_{x_1}} = \frac{92}{383.5} = .240$

$\alpha_1 = 13° \, 24'$



(2) Rotor.

From previous calculations, it was estimated that $\xi_R \cong .068$, $\therefore n = 1.37$ again.

$M_{1\,rel} = \frac{\omega_1}{a_1} = .367 , \quad \therefore \frac{A^*}{A_1} = .608$

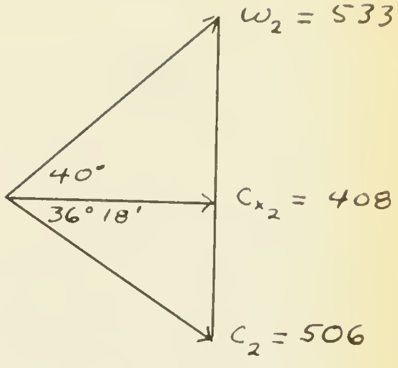
$\frac{A^*}{A_2} = .608 \frac{\cos 13.5°}{\cos 40°} = .771 , \quad \therefore M_{2\,rel} = .502$

$T_{02r} = T_{01r} = T_1 + \frac{\omega_1^{\,2}}{12,000} = 493°R$

$T_2 = \frac{T_{02r}}{1 + .2 M_{2r}^{\,2}} = 469° R$

$a_2 = 49.1\sqrt{T_2} = 1062$

$\omega_2 = M_{2r}\, a_2 = 533 \frac{ft}{sec}$



$$w_{u_2} = w_2 \sin 40° = 342$$

$$c_{x_2} = w_2 \cos 40° = 408$$

$$c_{u_2} = u - w_{u_2} = 300$$

$$\tan \alpha_2 = \frac{c_{u_2}}{c_{x_2}} = .735$$

$$\alpha_2 = 36°\text{-}18'$$

$$c_2 = \frac{c_{x_2}}{\cos \alpha_2} = 506$$

$$T_{02} = T_2 + \frac{c_2^2}{12,000} = 490.4 \text{ °R}$$

c. Work $= \dfrac{u \Delta c_u}{g J} = \dfrac{642\,(734-300)}{32.2 \times 778} = 11.1 \; \dfrac{Btu}{lb}$

3. Determination of loss coefficients. (See Fig. 10)

a. Nozzle.

$\theta^* = 62.4°, \quad \therefore \zeta^* = .066$ for normal blade

(1) Aspect ratio correction:

$$\zeta = (1 + \zeta^*)\left(.975 + .075 \frac{b_x}{l}\right) - 1$$

$$= 1.066\left(.975 + .075 \frac{.72}{1.55}\right) - 1 = 1.076 - 1$$

$$\zeta = .076$$

(2) Reynolds number correction:

$$R_e = \frac{D_h c_1 \rho_1}{\mu_1}, \qquad \text{use } \frac{p_1}{p_{01}} = \left(\frac{T_1}{T_{01}}\right)^{\frac{n-1}{n}}$$

$$p_1 = 2035 \times \left(\frac{480}{537}\right)^{3.7} = 1410 \; \frac{lb}{ft^2}$$

$$\rho_1 = \frac{p_1}{RT_1} = \frac{1410}{53.3 \times 480} = .0551 \; \frac{lb}{ft^3}$$

$$R_e = \frac{.0419 \times 828 \times .0551}{114.4 \times 10^{-7}} = 167,000$$

$$\xi = \left(\frac{100{,}000}{R_e}\right)^{.25} \xi^* = \left(\frac{1}{1.67}\right)^{.25} x .076 = .0664$$

No inlet angle correction was made to the stator.

$$\therefore \; \xi_{stator} = .0669$$

$$\eta_{cp} = \frac{1}{1.0664} = .938$$

$n = 1.37$ as used.

b. Rotor.

$\theta^* = 60°$, $\therefore \xi^* = .057$ for thin blade.

(1) Aspect ratio correction:

$$\xi = (1 + \xi^*)\left(.475 + .075 \frac{b_x}{\ell}\right) - 1$$

$$\xi = .081$$

(2) Reynolds number correction:

$$R_e = \frac{D_{hr}\, w_2\, \rho_2}{\mu_2}$$

for $n = 1.37$, $p_2 = p_1 \left(\frac{T_2}{T_1}\right)^{\frac{n-1}{n}} = 1410 \left(\frac{464}{480}\right)^{3.7}$

$p_2 = 1293 \frac{lb}{ft^2}$, $\rho_2 = \frac{p_2}{RT_2} = .05175 \frac{lb}{ft^3}$

$$R_e = \frac{.0768 \times 533 \times .05175}{112.4 \times 10^{-7}} = 188{,}000$$

$$\xi = \left(\frac{1}{1.88}\right)^{.25} x .081 = .0689$$

(3) Inlet angle correction:

$$\alpha_{1r} - \alpha_{1r}^{*} = -36°18' + 20° = -16°18'$$

$$K = 1.003$$

$$\xi = K(1+\xi^{*}) - 1 = 1.003\,(1.0684) - 1$$

$$\xi_{Rotor} = .07 \qquad \text{Check } n: \; \eta_{ep} = \frac{1}{1.07} = .935$$

$$n = 1.368$$

4. Determination of turbine efficiency.

$$\eta = \frac{.96\,W}{W + \left(\Delta h_{LR} + \frac{T_2}{T_1}\,\Delta h_{LS}\right)} \qquad (\text{Ref. } d.)$$

$$\Delta h_{LR} = \xi_R \times \frac{1}{2}\,\frac{\omega_2^{2}}{gJ} = .396\ \frac{Btu}{lb}$$

$$\Delta h_{LS} = \xi_S \times \frac{1}{2}\,\frac{c_1^{2}}{gJ} = .916\ \frac{Btu}{lb}$$

$$\frac{T_2}{T_1}\,\Delta h_{LS} = \frac{469}{480} \times .916 = .896$$

$$\therefore\ \eta_t = \frac{11.1 \times .96}{11.1 + .396 + .896} = 86.0\,\%$$

5. Check on pressure ratio.

$$\eta_t = \frac{1 - \frac{T_{02}}{T_{00}}}{1 - \left(\frac{P_{02}}{P_{00}}\right)^{.286}}, \qquad 1 - \frac{P_{02}}{P_{00}} = \frac{1 - \frac{490.4}{537}}{.86}$$

$$= .1022$$

$$\frac{P_{02}}{P_{00}} = (1 - .1022)^{3.5} = .685$$

$\frac{P_{00}}{P_{02}} = 1.46$ to compare with observed $\frac{P_{00}}{P_{02}} = 1.46$

APPENDIX C

Symbols

a	Sonic velocity, $\frac{ft}{sec}$
A	Area
AR	Aspect Ratio
b_x	Axial chord
c	Absolute velocity, $\frac{ft}{sec}$
D_h	Hydraulic diameter, for use with Soderberg Curves
h	Enthalpy, $\frac{Btu}{lb}$
k	Ratio of specific heats, $\frac{c_p}{c_v}$
K	Loss coefficient factor
l	Blade length
m	Mass flow, $\frac{lb}{sec}$
M	Mach Number
n	Polytropic exponent
N	RPM
p	Pressure
R_e	Reynolds Number
T	Temperature, degrees Rankine
u	Blade speed, $\frac{ft}{sec}$
w	Relative velocity, $\frac{ft}{sec}$
W	Work, $\frac{Btu}{lb}$
α	Air angle, degrees
β	Blade angle, degrees
η	Efficiency
ρ	Density, $\frac{lb}{ft^3}$

APPENDIX C

(cont.)

μ	Viscosity, $\frac{\text{lb.sec}}{\text{ft}^2}$
θ^*	Nominal angle of turn
ξ	Turbine loss coefficient

The use of 0 as a subscript prefix, indicates total temperature or pressure, eg:

p_{o1} - total pressure at point 1

The superscript * indicates either

1. Nominal (used with Soderberg Curves) or
2. Value of the quantity when the Mach Number is brought to unity.

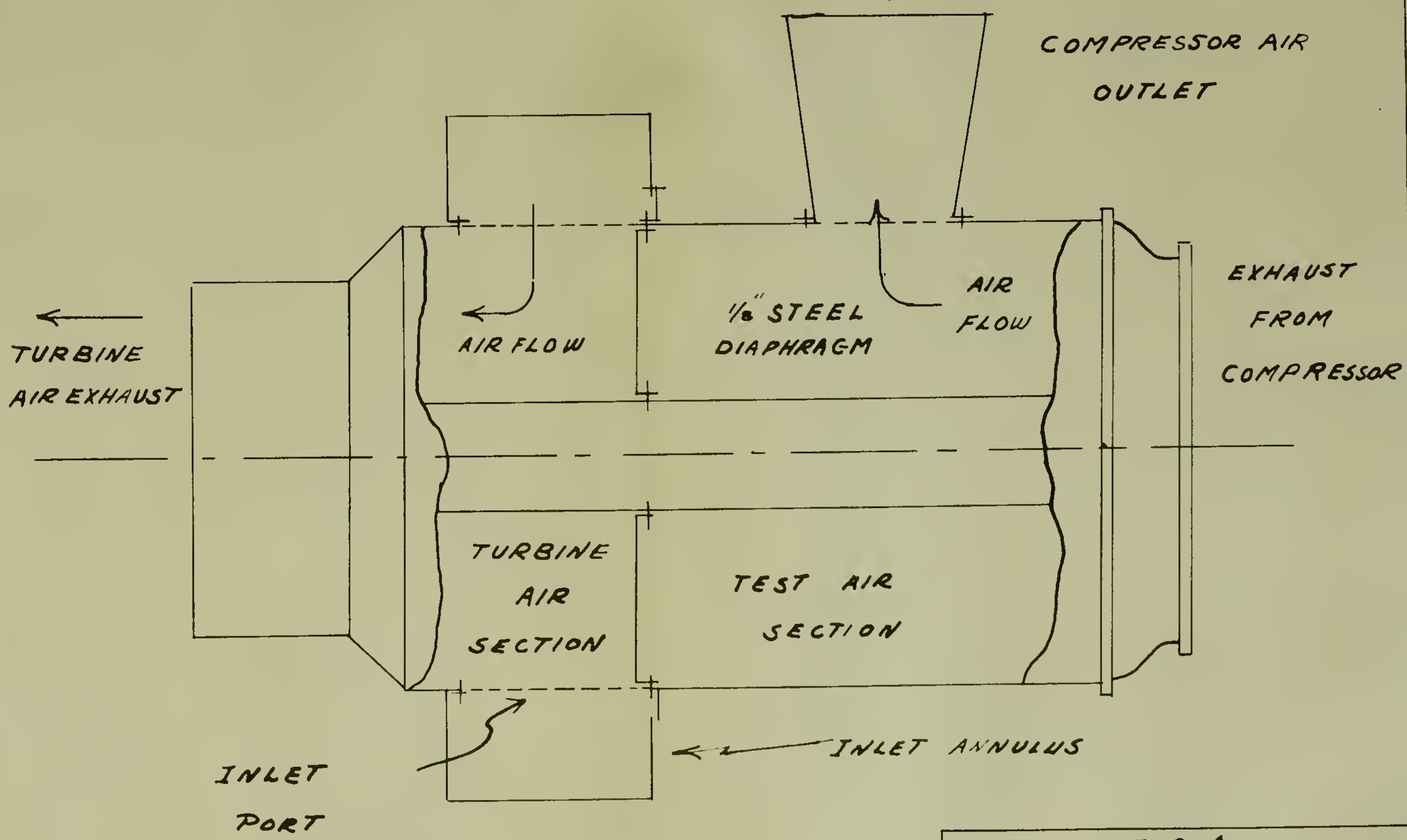


FIG. 1
SCHEMATIC SIDE VIEW
DUPLEX CHAMBER
INTERNAL ARRANGEMENT

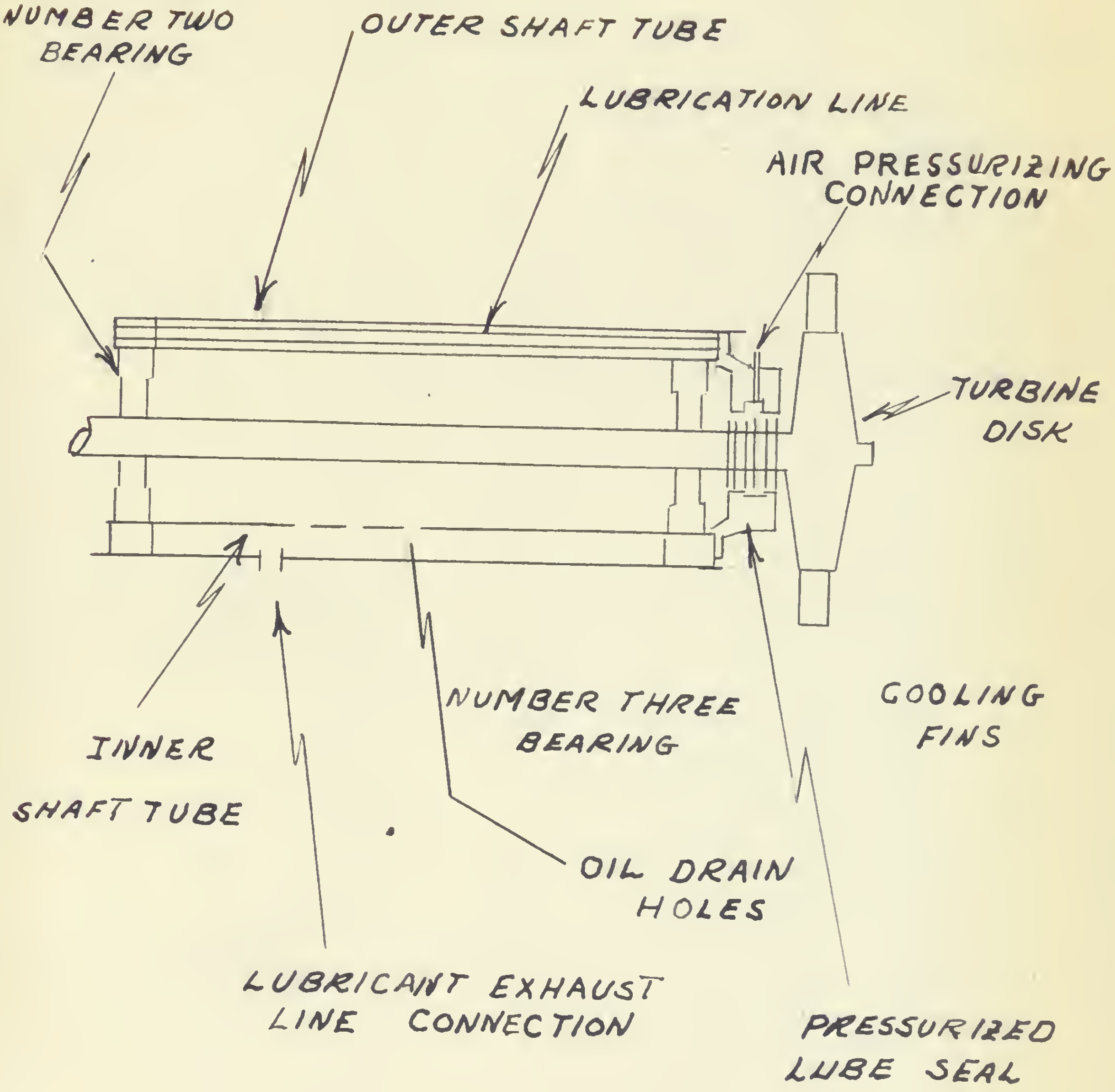


FIG. 2

SCHEMATIC SIDE VIEW
ALTERATIONS.
LUBRICATION SYSTEM

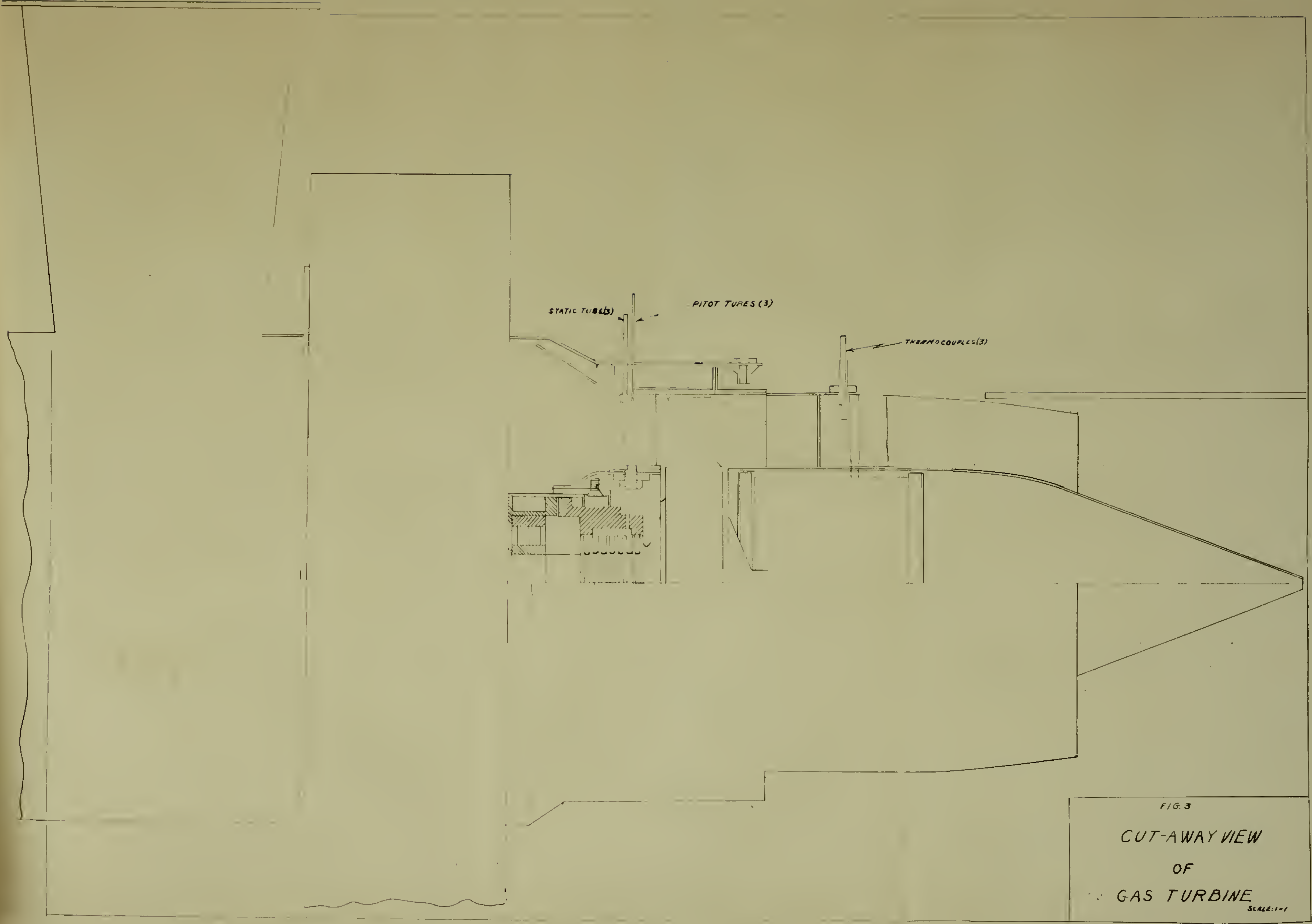
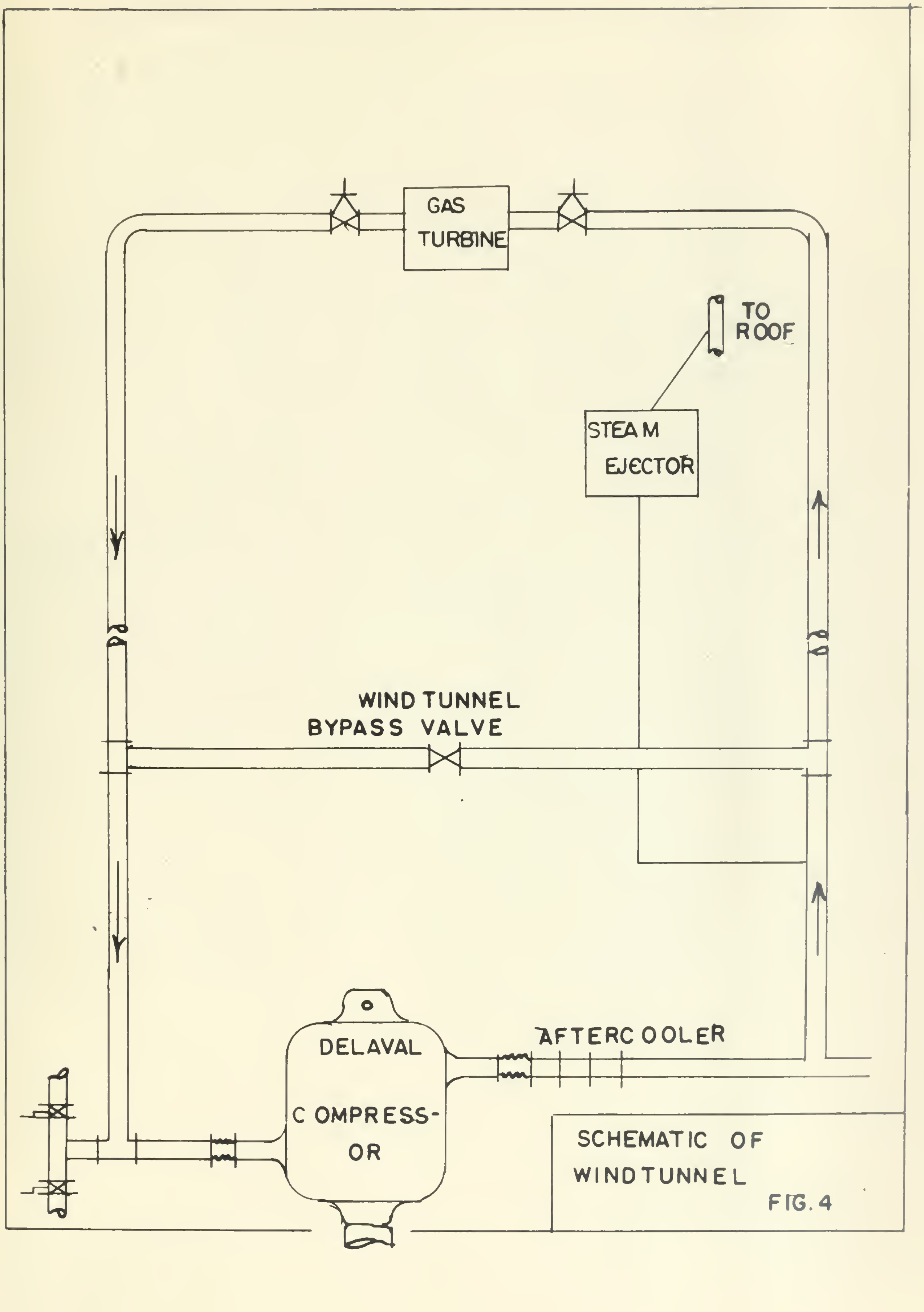


FIG. 3

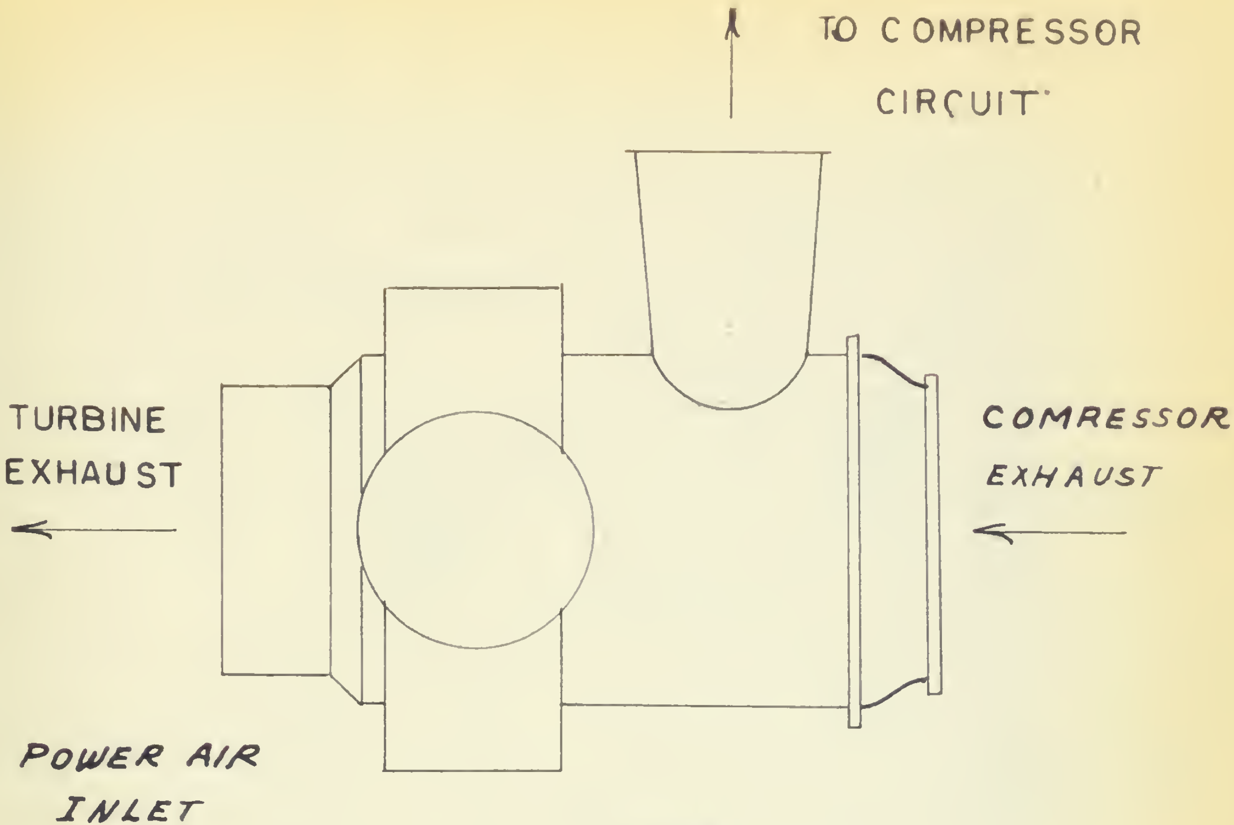
CUT-AWAY VIEW
OF
GAS TURBINE

SCALE:1-1

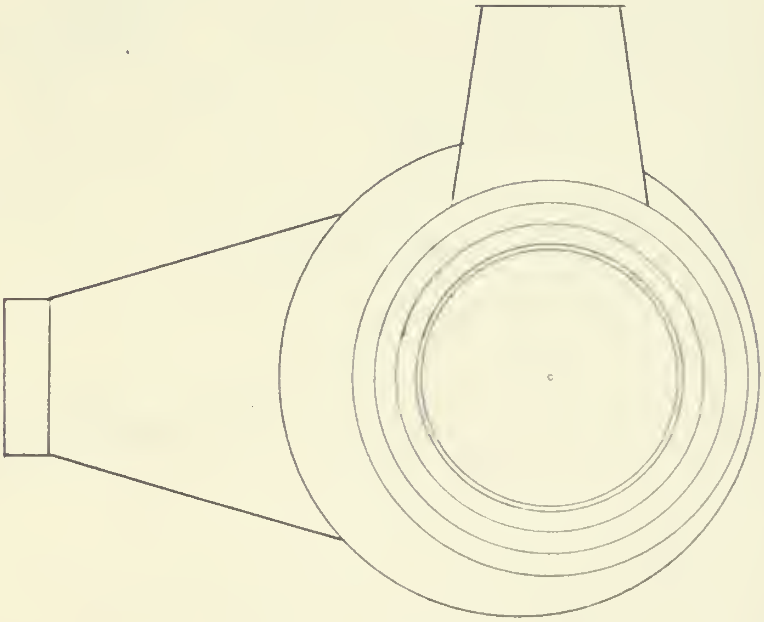


SCHEMATIC OF WINDTUNNEL

FIG. 4



SIDE VIEW



END VIEW

FIG. 5
DUPLEX CHAMBER
SCHEMATIC VIEW
EXTERNAL ARRANGEMENT

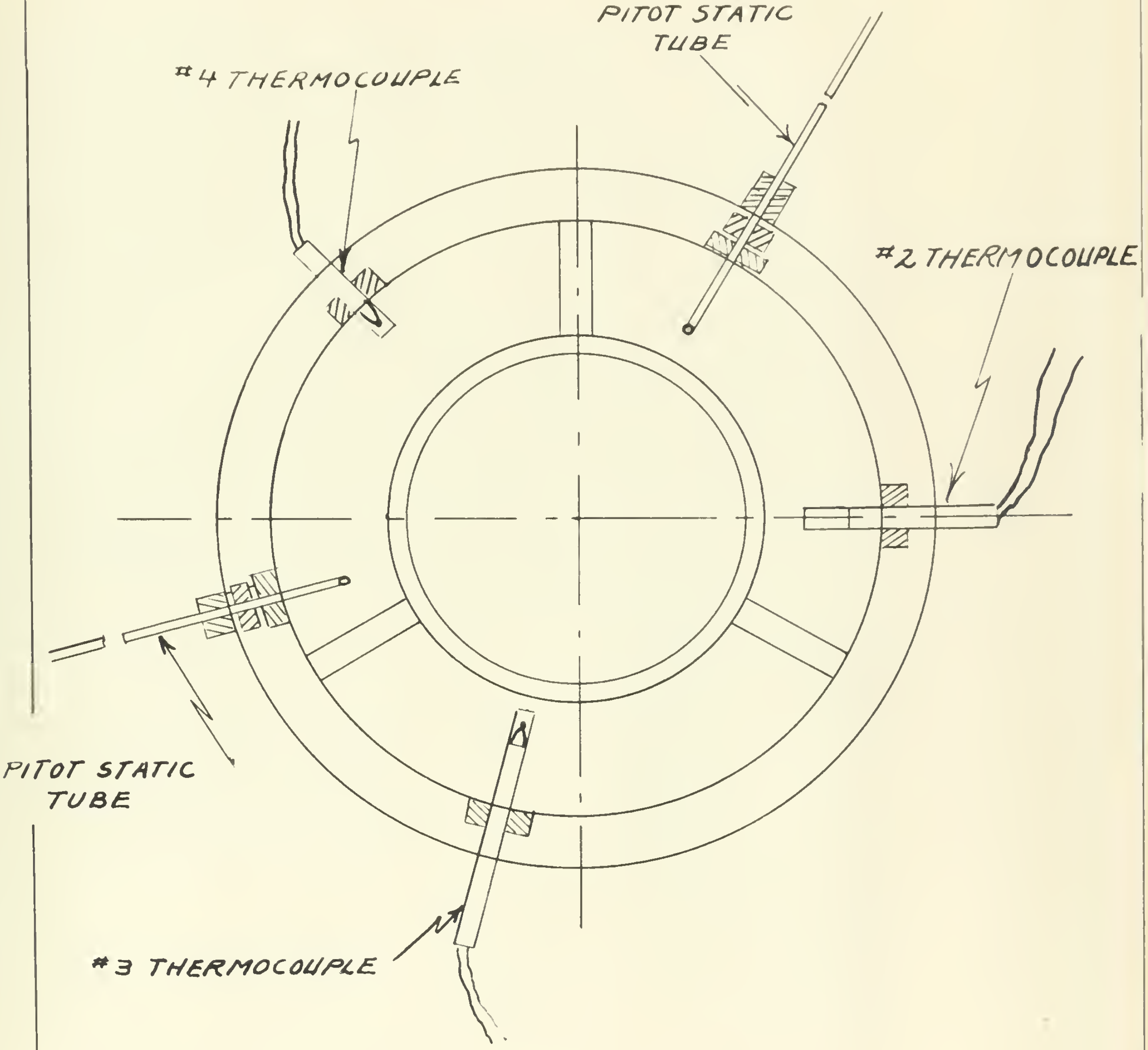


FIG. 6

AXIAL VIEW OF

OUTLET INSTRUMENTATION

SCALE 2:1

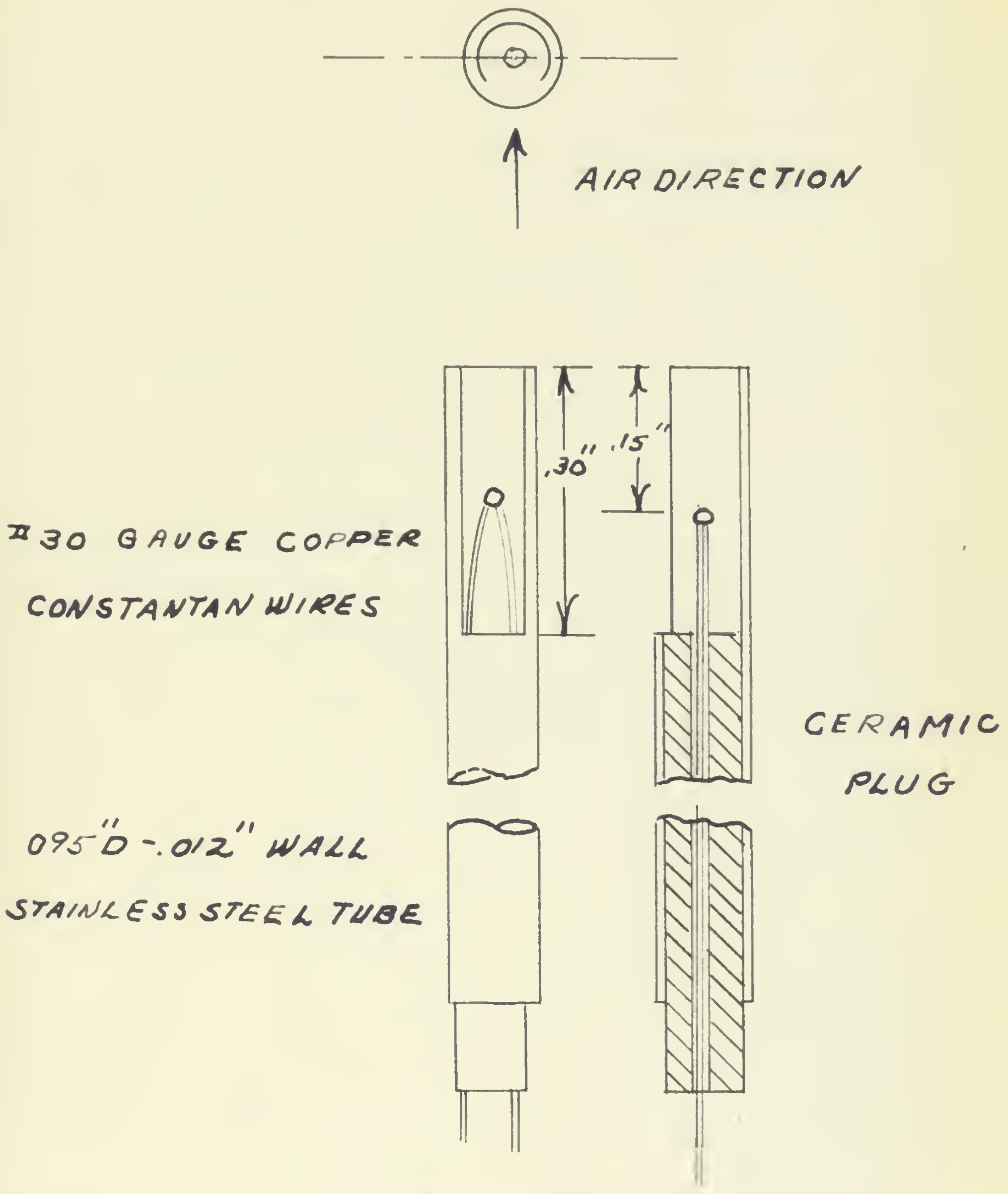

AIR DIRECTION
.30"
.15"
#30 GAUGE COPPER CONSTANTAN WIRES
CERAMIC PLUG
.095"D - .012" WALL STAINLESS STEEL TUBE

FIG. 7 TOTAL TEMPERATURE PROBE
CYLINDRICAL - HALF SHIELDED

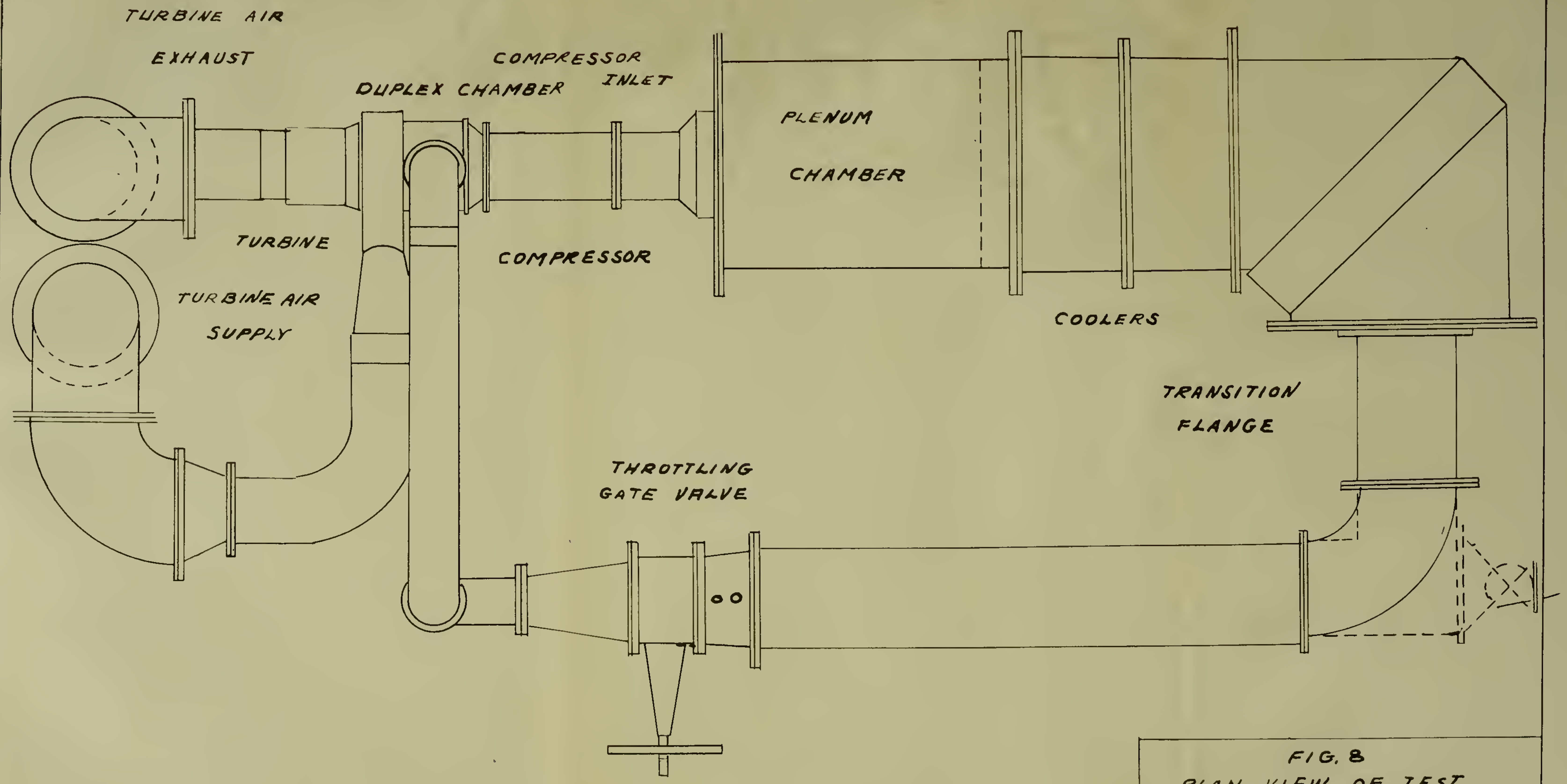


FIG. 8
PLAN VIEW OF TEST EQUIPMENT
SCALE: 1" = 1.0'

Polytropic Flow of a Perfect Gas with k = 1.4

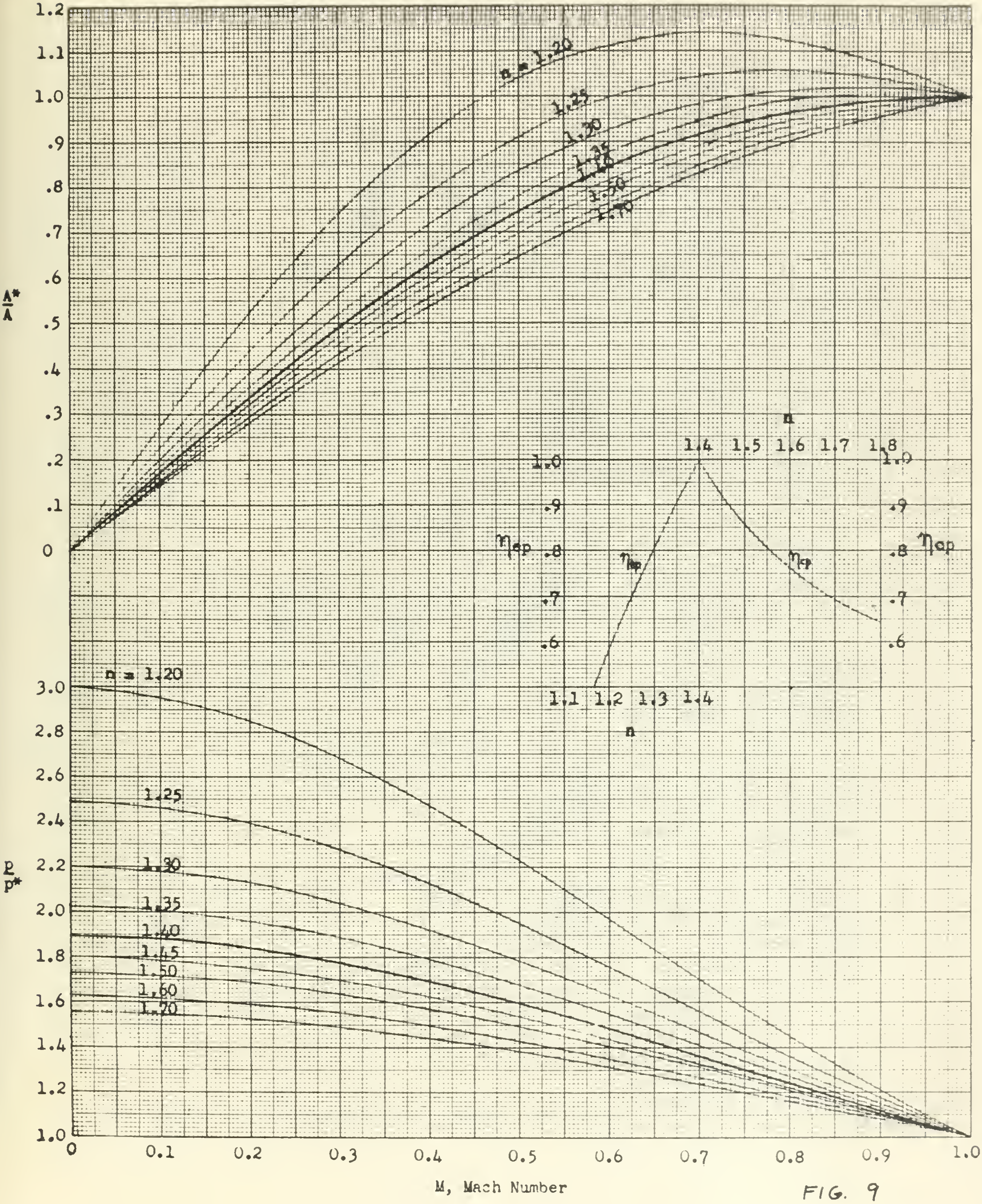
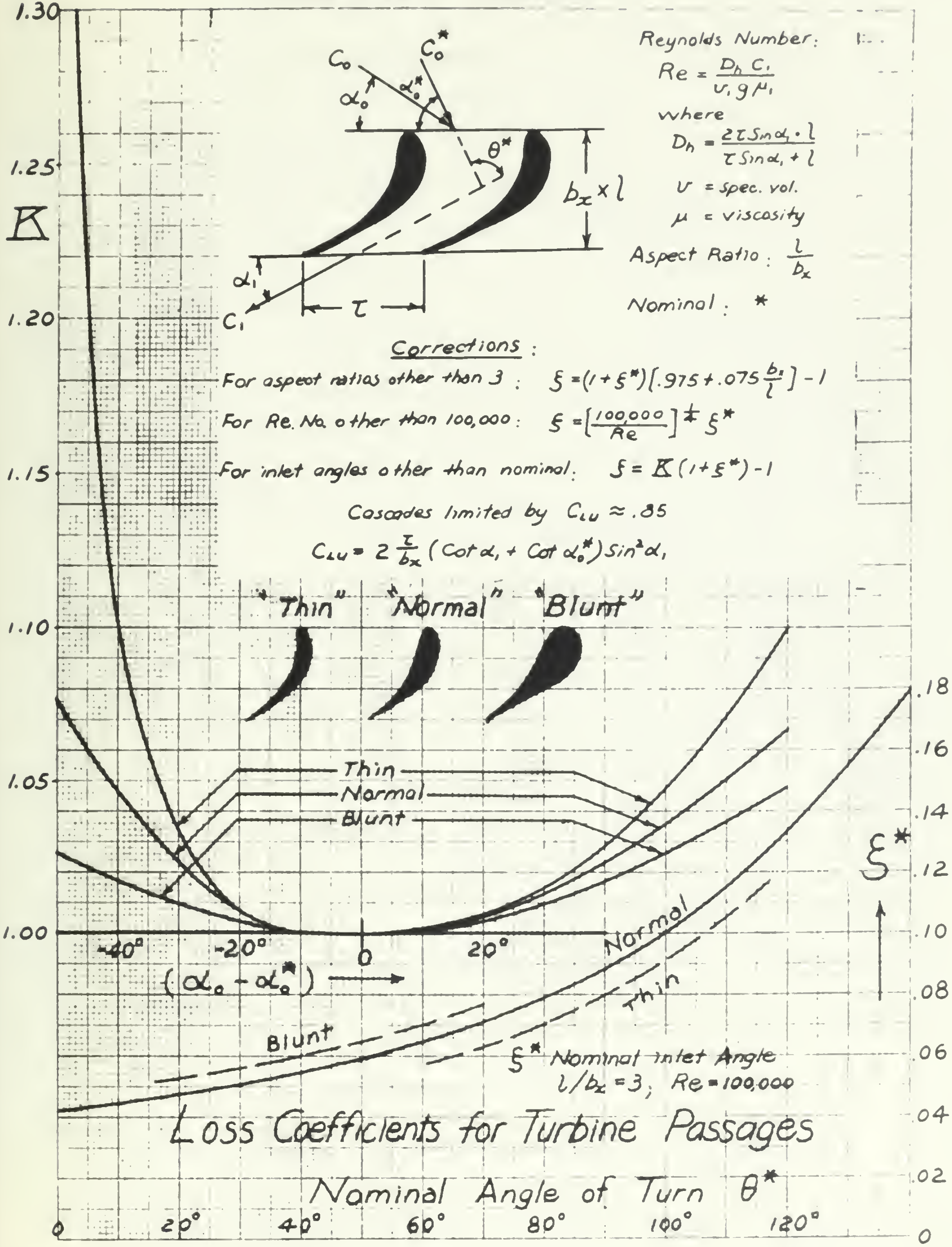


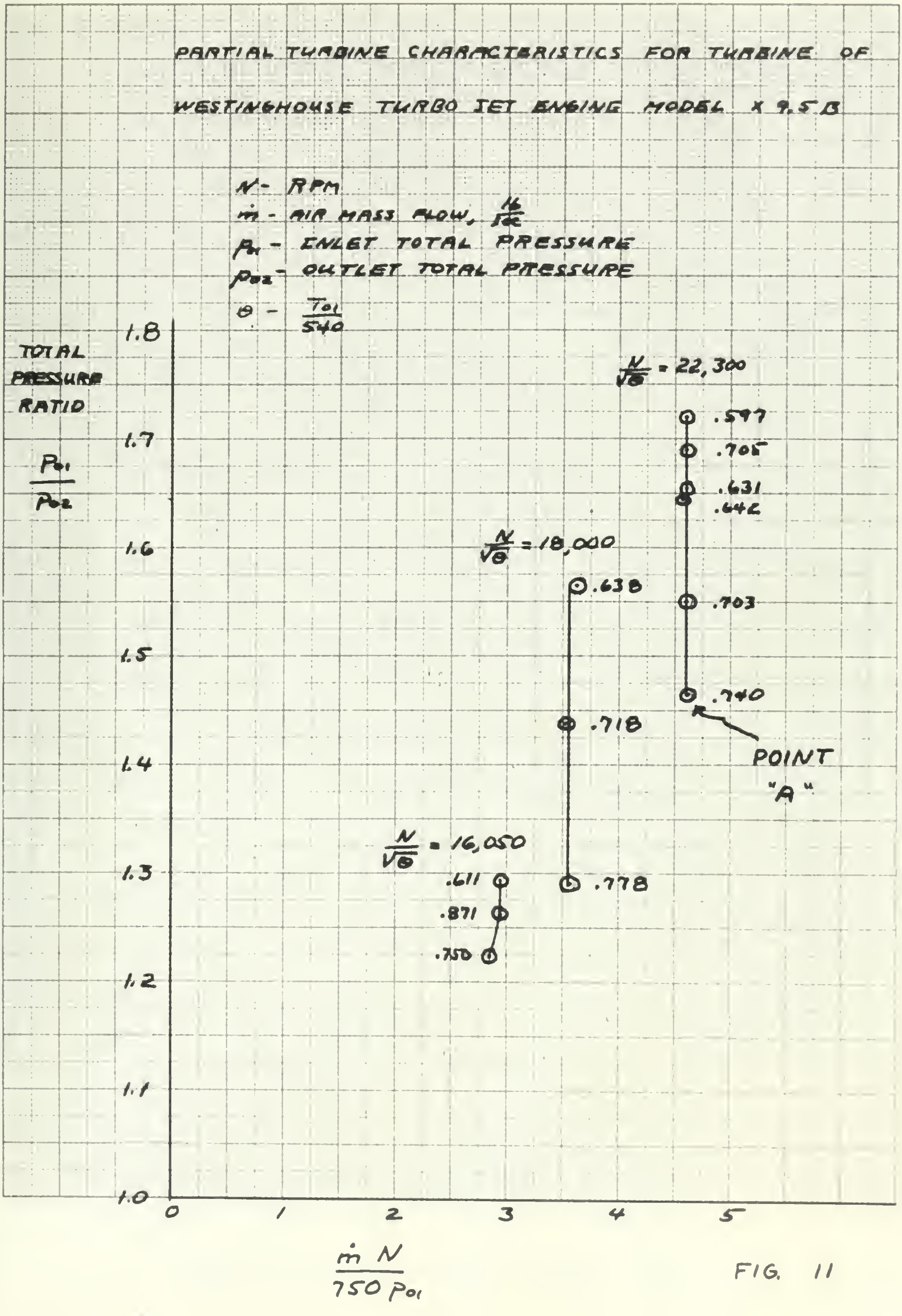
FIG. 9

SODERBERG CURVES



CRS - HY-1949

FIG. 10



PARTIAL TURBINE CHARACTERISTICS FOR TURBINE OF WESTINGHOUSE TURBO JET ENGINE MODEL X 9.5 B

N - RPM

$\dot{m}$ - AIR MASS FLOW, $\frac{lb}{sec}$

P_{01} - INLET TOTAL PRESSURE

P_{02} - OUTLET TOTAL PRESSURE

θ - $\frac{T_{01}}{540}$

FIG. 11

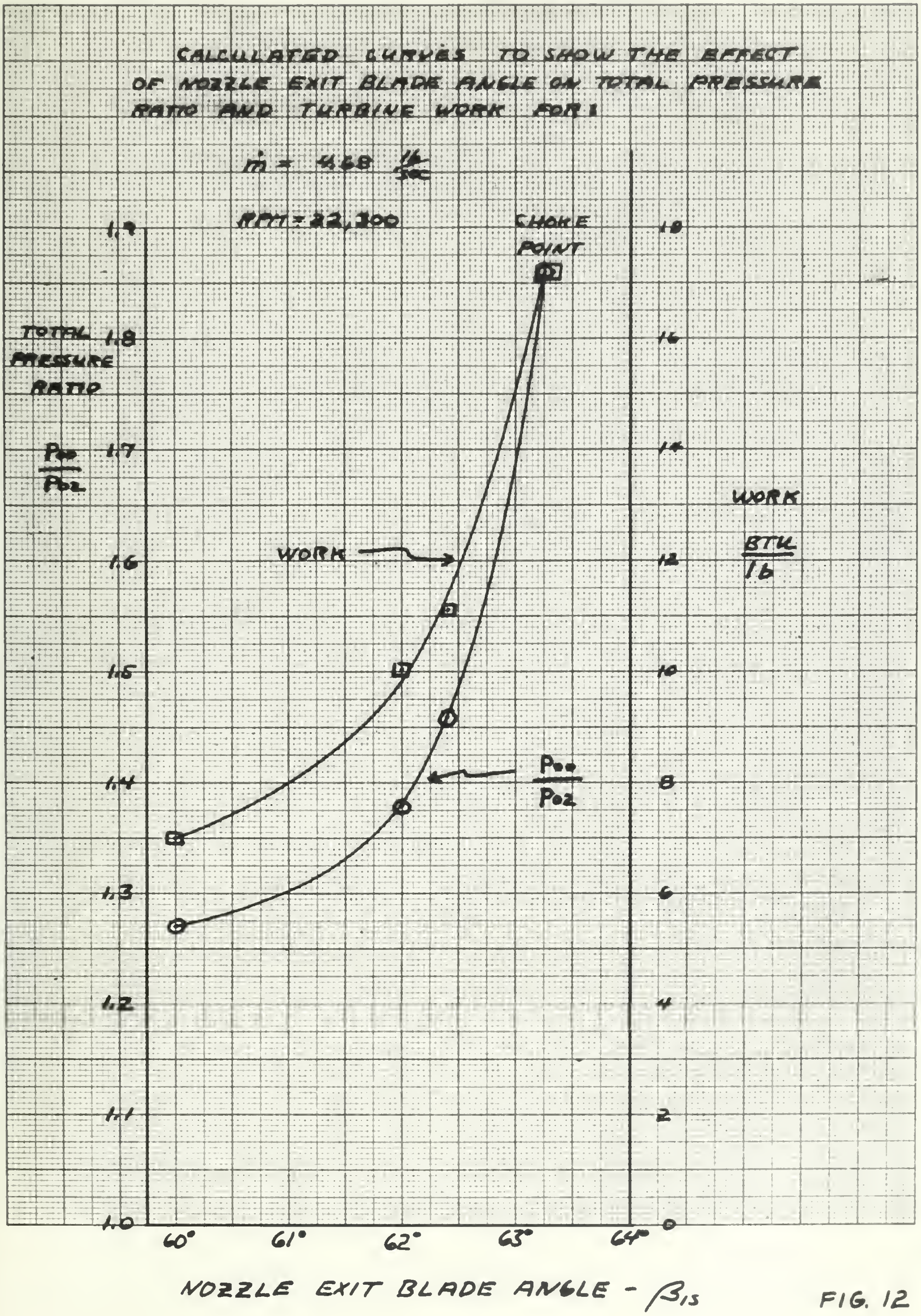

CALCULATED CURVES TO SHOW THE EFFECT OF NOZZLE EXIT BLADE ANGLE ON TOTAL PRESSURE RATIO AND TURBINE WORK FOR 1
ṁ = 4.68 lb/sec
RPM = 22,300
CHOKE POINT
TOTAL PRESSURE RATIO
P00/P02
WORK
BTU/lb
WORK
P00/P02
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
18
16
14
12
10
8
6
4
2
0
60°
61°
62°
63°
64°
NOZZLE EXIT BLADE ANGLE - β1s

FIG. 12

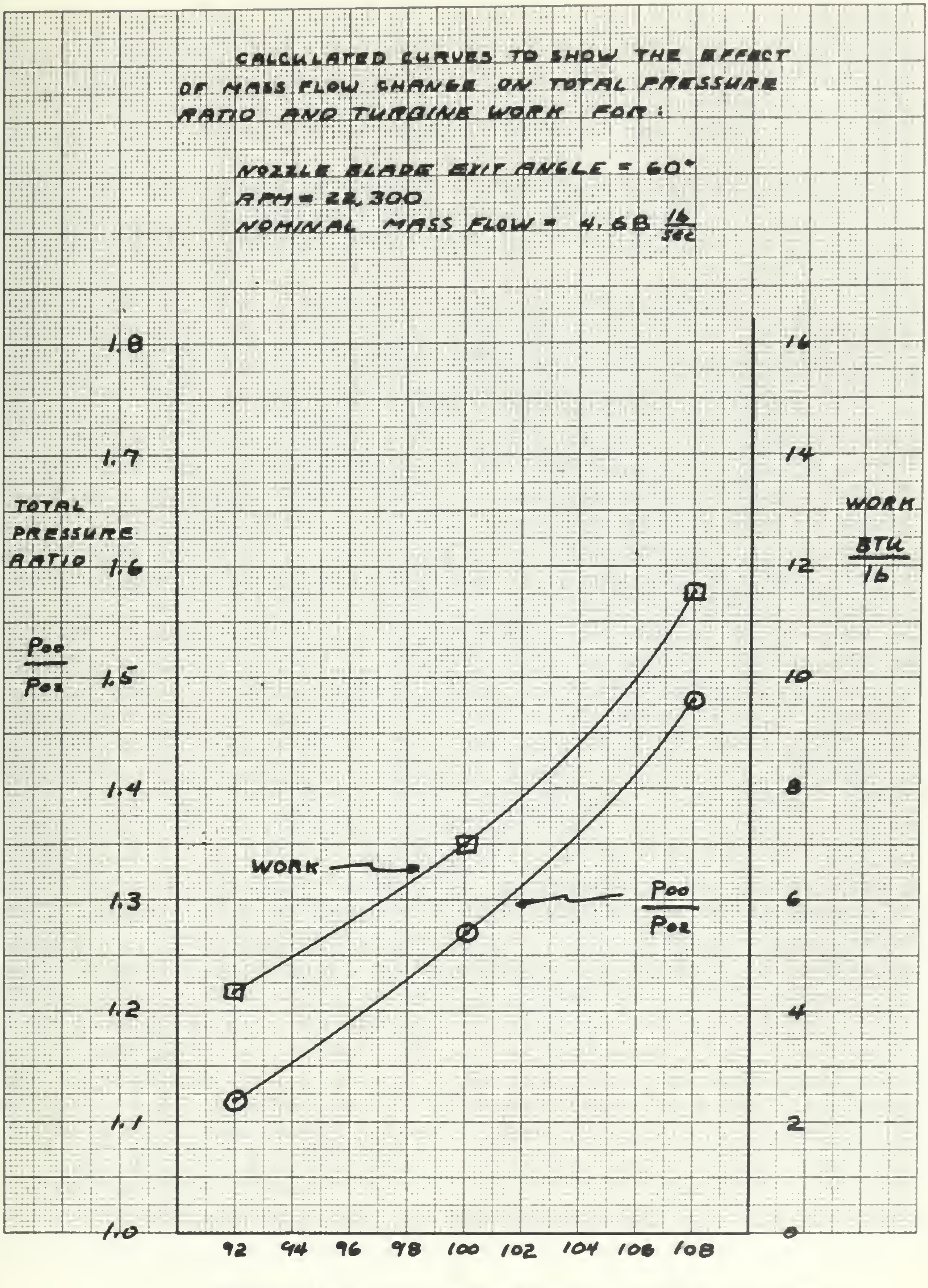

CALCULATED CURVES TO SHOW THE EFFECT OF MASS FLOW CHANGE ON TOTAL PRESSURE RATIO AND TURBINE WORK FOR:
NOZZLE BLADE EXIT ANGLE = 60°
RPM = 22,300
NOMINAL MASS FLOW = 4.68 lb/sec
TOTAL PRESSURE RATIO
$\frac{P_{00}}{P_{02}}$
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
WORK
$\frac{BTU}{lb}$
16
14
12
10
8
6
4
2
0
WORK
$\frac{P_{00}}{P_{02}}$
92
94
96
98
100
102
104
106
108
PERCENT OF NOMINAL MASS FLOW

FIG. 13

CALCULATED CURVES TO SHOW THE EFFECT
OF RPM CHANGE ON TOTAL PRESSURE RATIO
AND TURBINE WORK FOR:

NOZZLE BLADE EXIT ANGLE = 60°
MASS FLOW = 4.68 $\frac{lb}{sec}$
NOMINAL RPM = 22,300

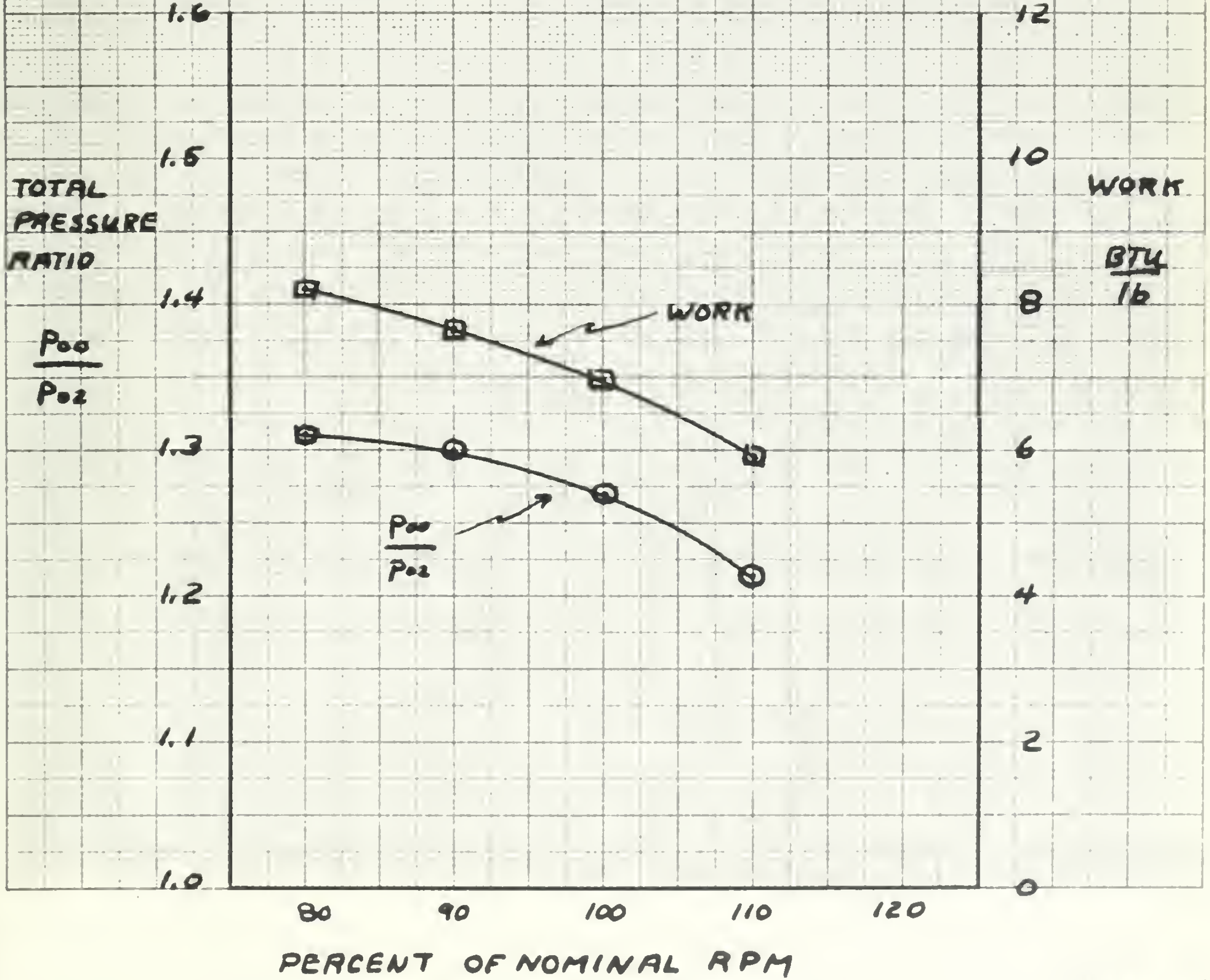


FIG. 14

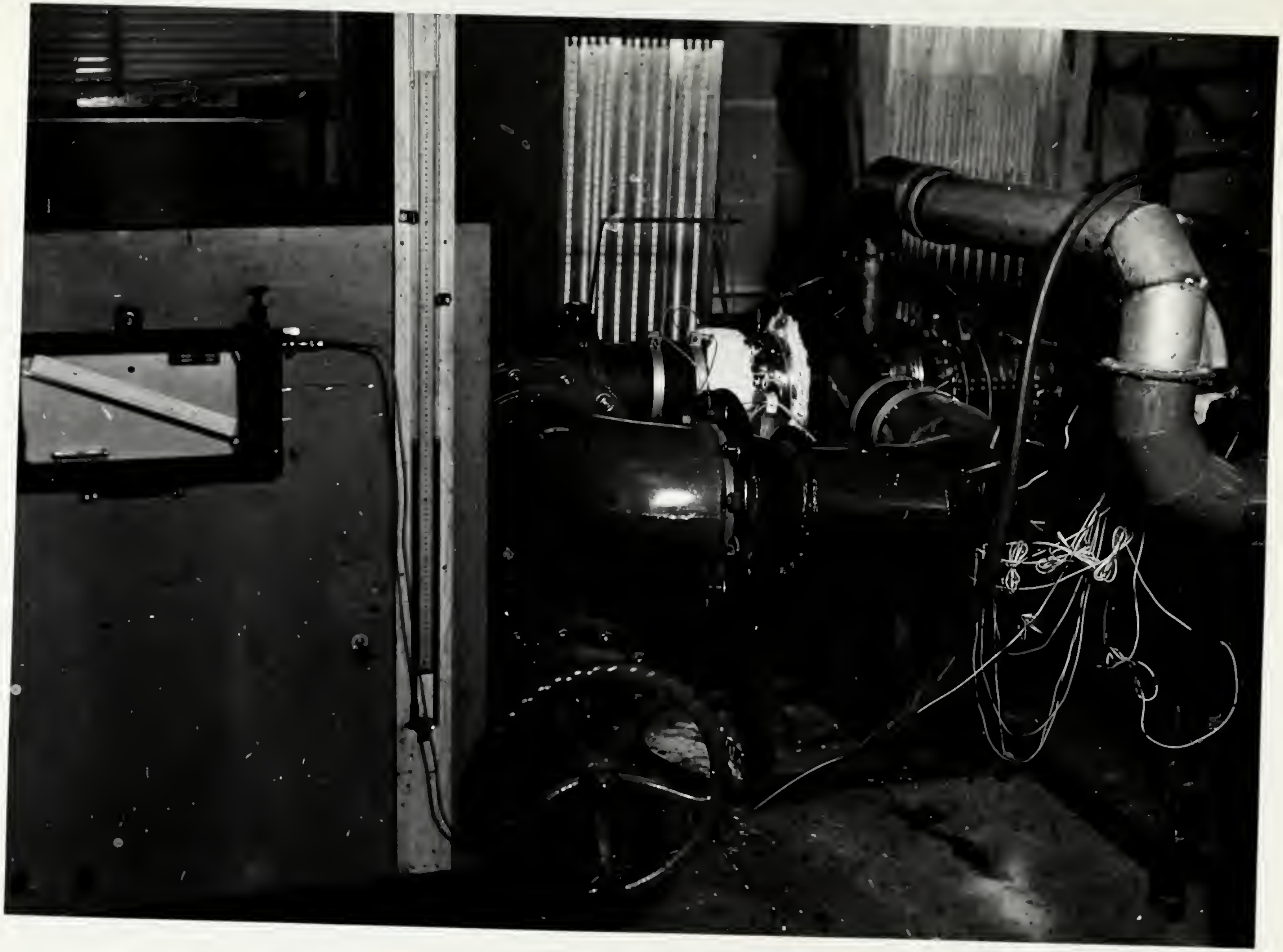


Fig. 15 - Side view of turbine and test apparatus. Manometer board for mass flow measurements shown at left.

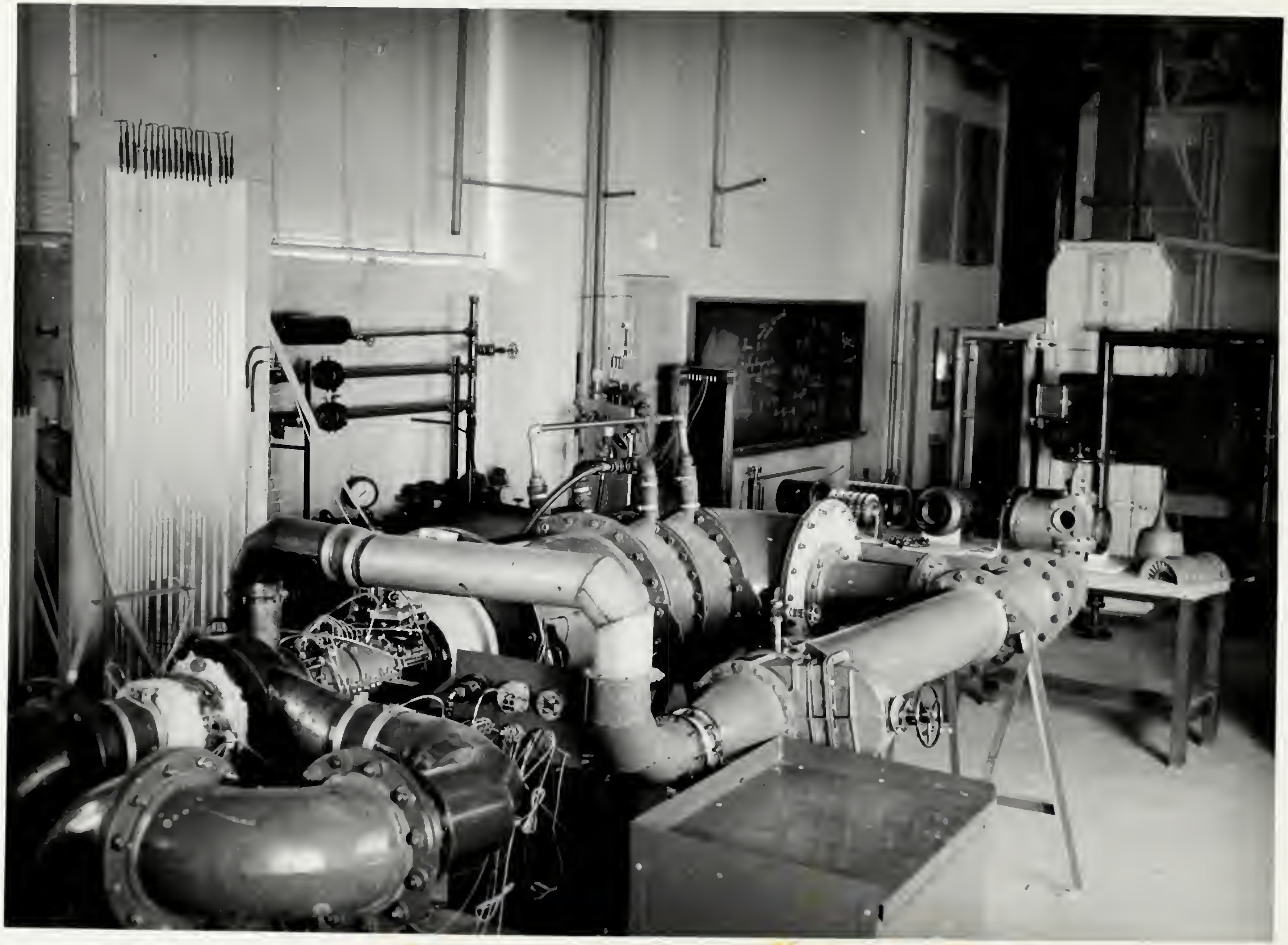


Fig. 16 - Overall view of turbine and compressor test apparatus.

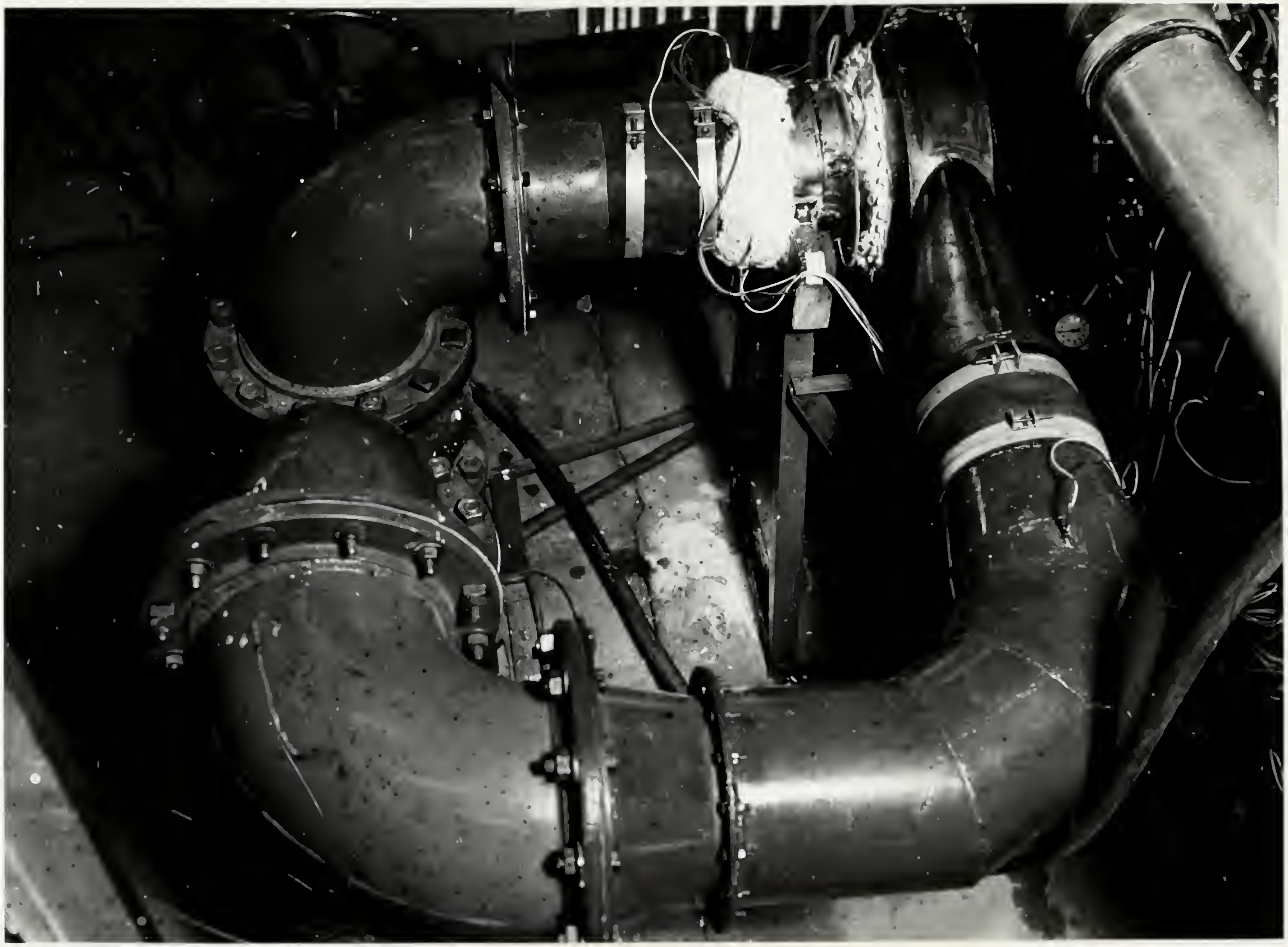


Fig. 17 - Overhead view of turbine test apparatus.

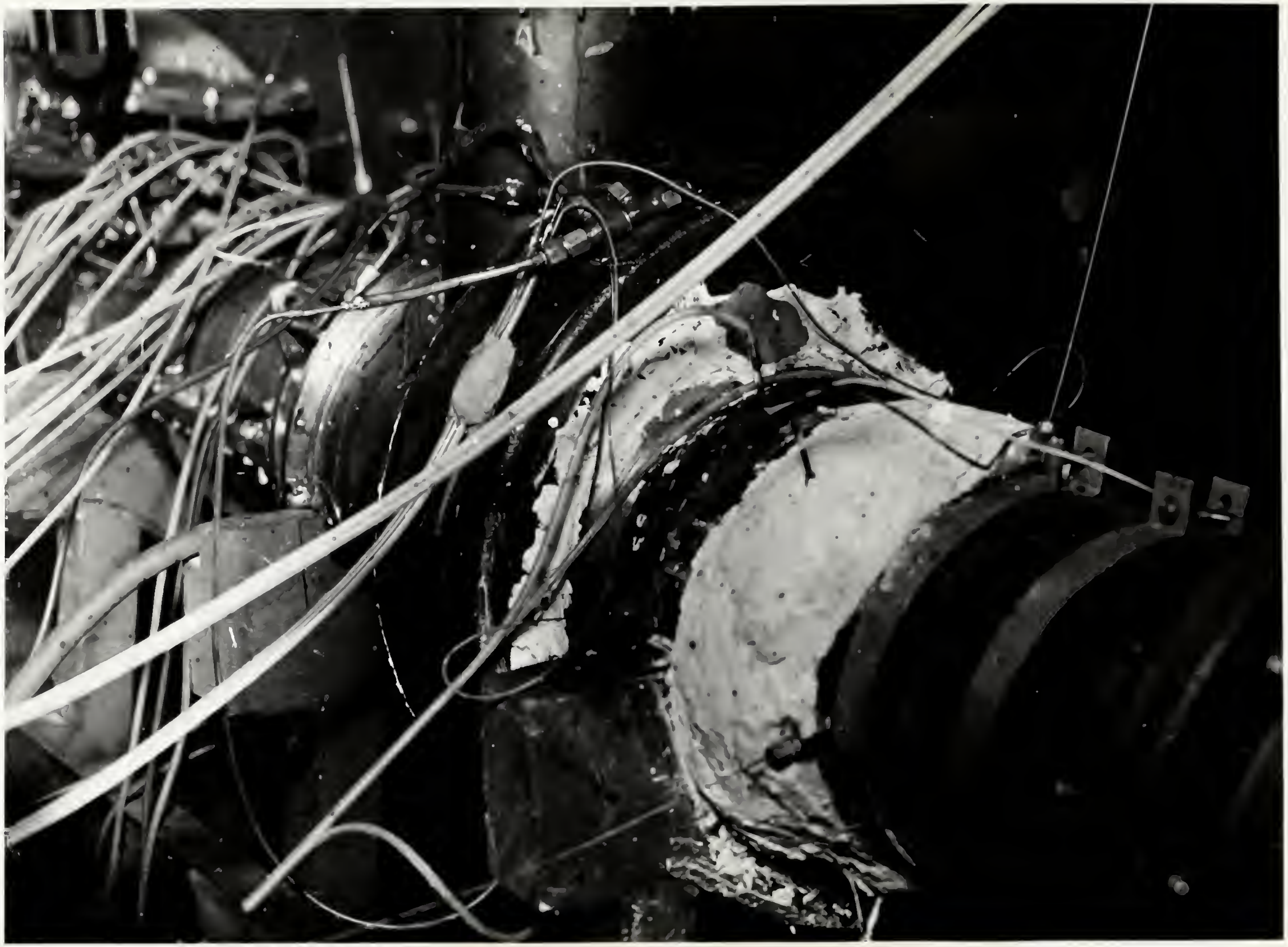


Fig. 18 - Three-quarter view of turbine test apparatus.

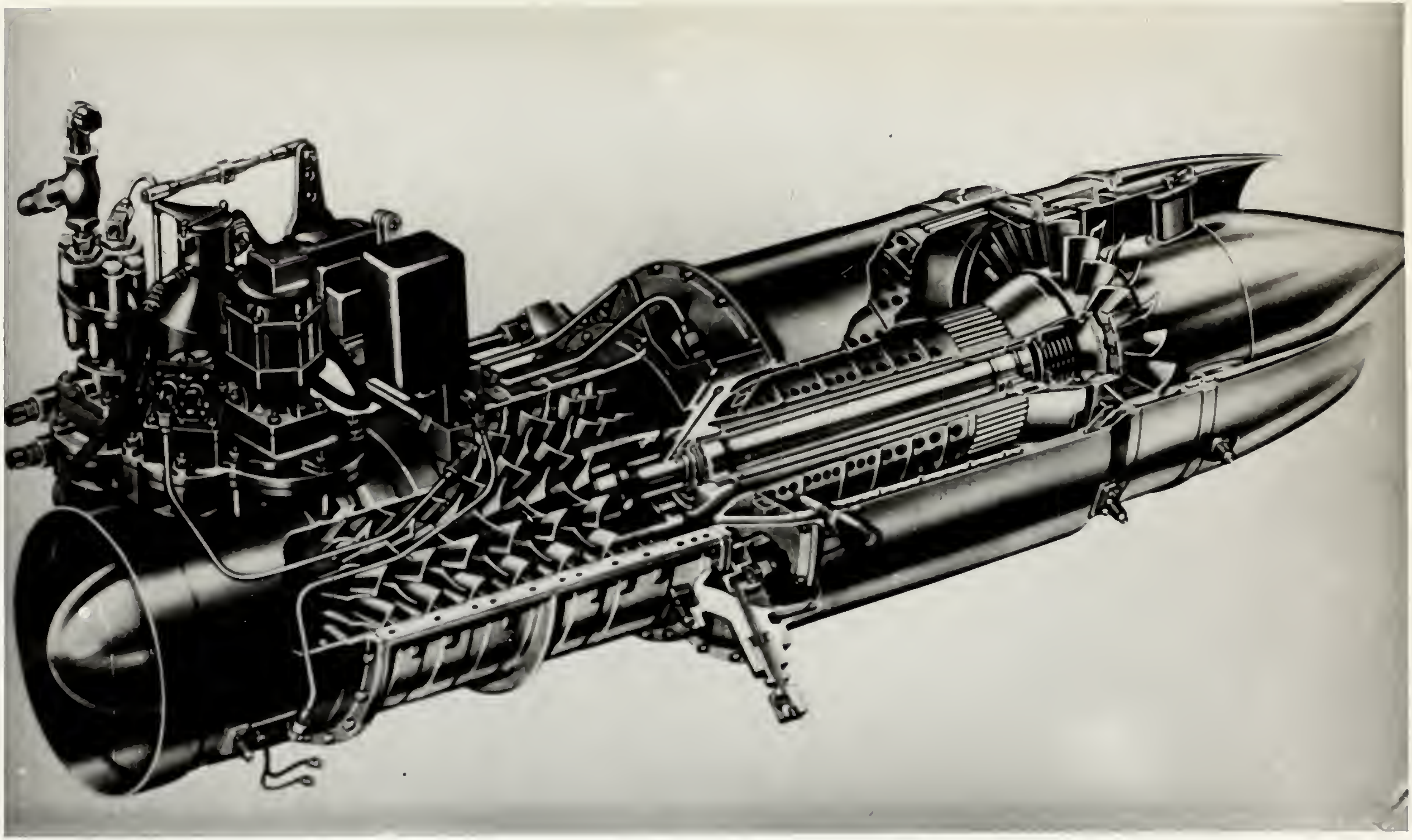


Fig. 19 - Westinghouse X9.5B jet engine.

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